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FINITE ELEMENT ANALYSIS OF
CANTILEVER VALVE REEDS IN RECIPROCATING GAS COMPRESSORS

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SYNOPSIS

Analyses of the dynamic behaviour of cantilever valve reeds in reciprocating gas compressors were made both theoretically and experimentally. Theoretical predictions were obtained using the Finite Element Method. Techniques were applied which minimised the necessary computer resources. Conventional experimental techniques were extended to determine the forced dynamic displacements and strains. Predicted dynamic displacements and strains were in acceptable agreement with experimental results.

INTRODUCTION

In small reciprocating compressors the valves usually consist of flexible steel plates (reeds) clamped at one end and covering one or more ports. To protect the reeds from over-stressing, the valve lift is limited, either by a point stop or by a backing plate. The resulting non-linear boundary conditions complicate the mathematical model required to simulate the dynamic behaviour of the reeds.

Techniques to study the behaviour of valve reeds under operating conditions have advanced during the last decade (1,2,3,4). Usually the valve reed has been described as a single degree of freedom spring-mass system: such a simple description does not provide adequate information concerning stress patterns. The complexity both of the geometry of valve reeds and of the non-linear boundary conditions favour analysis by the Finite Element Method (5,6,7,8).

PROGRAMS FOR ANALYSIS BY THE FINITE ELEMENT METHOD

A suite of programs* was developed:

- a) "GRID": a program, after Segerlind (9), to generate the mesh grid automatically. The geometry of the broad regions of a structure is

specified (via a "conversational" mode) by the global x,y co-ordinates of the input nodal points and by the connectivity data (topology) of these regions. At input, eight-noded quadrilateral elements are employed and the number of rows and columns per region is specified. The computer plots this initial geometry of the structure and then generates quadrilateral elements for the finite element model. The nodal x,y co-ordinates of this model are calculated and then each quadrilateral element is divided into two triangles by a hypotenuse: the smaller hypotenuse is used in order to give a better aspect ratio. The resulting finite element grid with triangular elements is plotted (19). The bandwidth of the stiffness matrix is estimated (but not minimised).

- b) "DTCHK": a program to check input data. This program (19) is designed to reveal pictorially any error in the specification of the geometry of the grid. The x,y co-ordinates of the grid are read and then the grid is plotted. A number is assigned to each element and plotted inside it showing that each element exists and is not merely formed by the surrounding elements.

Errors in the specification of the grid geometry can be identified, at some expense of computer resources, by the main suite of the finite element programs ("DRST4", "RESP95ST") but only when the errors result in a stiffness matrix which is non-positive definite. The location of errors is not specified. Moreover, any errors which cause an artificial hole or small overlapping of elements may result in the absence of the prior condition of a non-positive stiffness matrix.

- c) "MINBAND": a program, after Collins (20), to assign revised node numbers to the finite element model in such a way as to reduce the bandwidth of the stiffness matrix in order to economise on computer resources (19,20). At input, the relationship of the location of the nodes to each other is specified. The bandwidth of the stiffness matrix is calculated and

* A listing of these programs is available on Inter Library Loan from the Librarian, University of Strathclyde, Glasgow, quoting this paper and Accession Number 055/2477/9 as reference.

then reduced by starting the numbering of the nodes in the area of greatest flexibility (at the largest distances from the clamped boundary).

- d) "DRST4": a finite element program using the direct stiffness method to analyse the static, the free vibration and the instability characteristics of structures. This program, by Soper (10) of the University of Strathclyde, consists of seven separate but interdependent programs (11). The major operations are: evaluation of element property matrices, assembling them into the overall stiffness matrix, solution of the stiffness equation and then evaluation of stresses and accumulated loads. An attempt is made to minimise input data and to be efficient in the use of computer core capacity, both in method of solution and in the structure of the program (7). The Cholesky decomposition method is used in the solution of the stiffness equation. The Givens Householder iteration technique is employed to retrieve the eigenvectors. Consistent formulation of the mass matrix is preferred to a lumped formulation. When evaluating the natural frequencies and mode shapes for systems with many degrees of freedom, an eigenvalue economiser version of "DRST4" may be employed in order to further save computer resources. Thus, a larger number of degrees of freedom could be permitted but with a penalty of some loss in accuracy in the prediction of the higher natural frequencies and mode shapes (7, 10, 19).

- e) "RESP95ST" (Response, 95 maximum degrees of freedom, stress): a program after Heyes (11), for dynamic analysis of structures with fixed boundary conditions (7, 19). The program is based on a direct integration (step-by-step) method and was used to analyse the dynamic behaviour of reeds, without limitation of lift, when subjected to a specified pressure-time history. In the integration scheme, linear variation of the acceleration per integration time interval was assumed. Numerical damping (19) was not included. Viscous damping was assumed by making the damping matrix proportional to the stiffness matrix. A 3-noded flexural triangular element and a 4-noded flexural rectangular element, each with 3 degrees of freedom per node (one translational and two rotational), were available. The 3-noded triangular elements employed in "DRST4" and "RESP95ST" had linear variation of internal stress or strain and satisfied internal but not boundary compatibility. The internal stress or strain relationships in the 4-noded rectangular elements were predominantly linear but did not involve terms as high as cubic. These elements satisfied internal compatibility but normal slope incompatibilities existed at the boundaries (10). Neither type of element satisfied internal nor boundary equilibrium. The deflection within the triangular element was described by a third degree polynomial: a 6 degree polynomial was used with the rectangular element. Neither of the polynomials was

complete. Convergence criteria were applied: convergence was correct, rapid but not monotonic when rectangular elements were used. Test problems (10, 19) indicated that correct convergence was achieved with the triangular elements.

When rectangular elements were employed, relatively high values of natural frequencies were predicted. Refining the finite element grid did not improve accuracy. Addition of a third rotational displacement at each node could be expected to do so (11). Other modifications to program "RESP95ST" might be considered. For example, introduction of an economiser facility (master/slave) into this program could increase the number of degrees of freedom which it was practical to adopt. Inclusion of "higher order elements" may have beneficial effects where "curved" boundaries are involved.

- f) "RBCRST": a program used in conjunction with "RESP95ST", mainly to account for the non-linear boundary conditions imposed by the valve seat and stop. The stop may be a point stop, as is usual with cantilever suction reeds, or a backing plate, as often used in discharge valves. The direct integration scheme used in "RESP95ST" was only conditionally stable. "RBCRST" provided unconditionally stable integration schemes, based on the Newmark or Wilson- θ method (7), which were more versatile and in many cases more accurate. In such schemes, numerical damping and several variations of acceleration in each integration time interval (constant, step functions, linear or otherwise) can be employed. The time interval used in unconditionally stable integration may be long and may result in inaccuracies in the higher frequencies predicted for the system (19). Numerical damping may be introduced to filter these frequencies. The assumed form of acceleration depends upon the physical characteristics of the system, the accuracy desired and the stability of the procedure (7, 19). A coefficient to account for oil stiction at valve seat and stop (19, 21), a coefficient of restitution to account for valve bounce at seat or stop and a coefficient of gas pressure drag on the reed (as a function of valve lift) were included.
- g) "RDDSP": a program used to read the dynamic displacement at nodes along a reed predicted by "RESP95ST" and "RBCRST" and then plot the reed motion (19).

NON-LINEAR REED BOUNDARIES

When a reed node violated the limits of displacement imposed by the valve seat or stop, the node velocity and acceleration were declared to be zero and the displacement of this node was declared equal to the displacement limit (19). This procedure avoided having to solve for changing boundary conditions when a reed struck the seat or the stop. Dynamic displacements and bending stresses were predicted well if small time increments were employed. However, there is no guarantee that

either stability or convergence will be achieved and each case has to be investigated. Calculations should be repeated using progressively small time increments and finer grids to ensure that sufficient accuracy has been achieved, although this increases the computer resources required. Jarvis (13) reported that the method was used successfully by Pafec Ltd. to handle similar problems. However, it was stated not to be applicable to all such situations and that instability might persist.

There are alternative methods to account for the non-linear boundary conditions. "Diode" non-linear elements could be employed. Alternatively, the matrices of the equation which governs the dynamic behaviour of the valve can be portioned to account for the known force excitation due to gas pressure loading, and displacement excitation due to valve seat or stop (14). However, this method makes high demands on computer resources.

In the case of a point stop, the program "RESP95ST" can be halted and the input and boundary conditions changed manually when the reed is about to leave or touch the stop (7). If the reed flutters, this procedure becomes very tedious. When changes in boundary conditions of this type are programmed, excessively high computer resources are required (13).

Hamilton (14) proposed to account for the non-linear boundary conditions when a reed touched a point stop by adding a very large value to the appropriate coefficient of the lead diagonal of the stiffness matrix. With this method, a matrix composed from the mass matrix and the revised stiffness matrix has to be inverted each time the boundary conditions alter, so increasing the computer resources required.

Elson et al (15) developed a non-linear, one dimensional model to describe the dynamic displacement of a half-annular reed with a backing plate. The non-linear boundary conditions were accounted for from an experiment in which the valve was vibrated against its backing plate in a bench test in order to measure the effective frequency function of the valve.

INTEGRATION TIME INCREMENT

The dynamic response of valve reeds with a point stop is described adequately by the first few modes. However, the numerical integration scheme employed introduces significant error in the period and amplitude of the contributing modes unless a sufficiently small time increment is used. The errors are small when the time increment used is about 6% of the period of the highest mode included (16). The existence of excited higher modes was predicted by the program "RBCRST" for a reed striking a backing plate or valve seat. Such behaviour could not have been predicted if an adequate number of mode shapes had not been allowed for in the model, even if sufficiently small time increments had been employed in the integration scheme. Experimental verification of these higher modes being excited has been provided by Woollatt (17).

REED CLAMPING

Valve reeds mounted in a compressor are imperfectly clamped, particularly when rubber clamp pads are used. The effect of imperfect clamping on the dynamic behaviour of reeds was accounted for by adding extra elements with variable stiffness and density at the root of the reed (7,11). Alternatively, more economical use of computer resources could have been achieved by appropriately increasing the coefficients corresponding to the nodes at the reed root in the lead diagonal of the stiffness matrix.

DYNAMIC BEHAVIOUR OF CANTILEVER REEDS

A schematic view of a modified cylinder head and valve plate is shown in Fig. 1. Displacements at points along the reed were obtained by a modified Wayne Kerr displacement transducer; a Kistler piezo-electric transducer measured the pressure-time history and miniature foil strain gauges (Showa, type N11-FA-05) recorded strain along the valve reed (8).

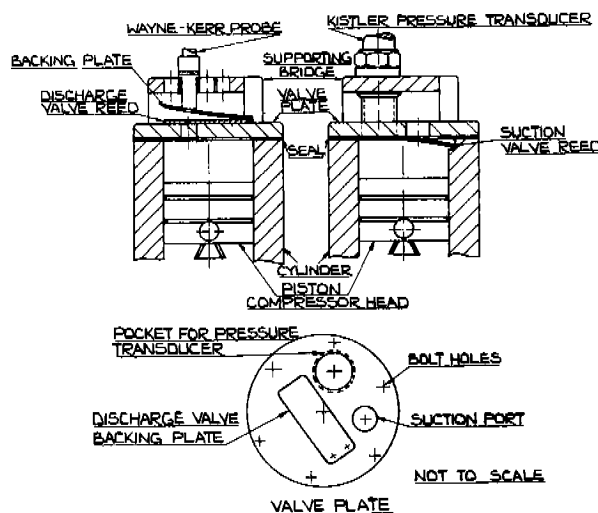


FIG. 1 SCHEMATIC VIEW OF MODIFIED CYLINDER HEAD AND VALVE PLATE OF TEST COMPRESSOR

Satisfactory agreement was obtained (Fig. 2) between dynamic displacements predicted by the Finite Element Method and those measured for a cantilever discharge reed with a backing plate. At the centre of the port the reed was observed to bounce after it reached the backing plate. This bounce was predicted by the analytical model even when the coefficient of restitution was set to zero, so the small oscillation was considered to be due to changes in the reed stiffness as it contacted the backing plate (15,17). Slight movement of the backing plate while the reed was pressed against it possibly explains the discrepancy between analytical and experimental displacements when the valve was fully open. Moreover, the finite element grid used was not fine enough to predict very slight distortion of the reed as it was pressed into the pocket containing the displacement transducer.

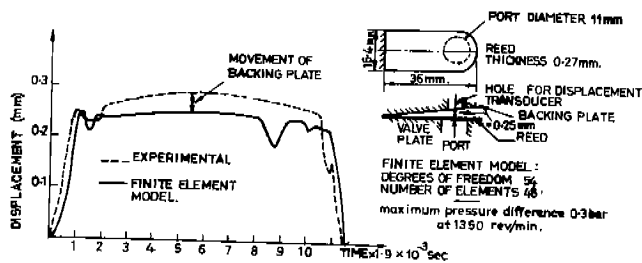


FIG. 2. DISPLACEMENT OVER THE PORT CENTRE OF A CANTILEVER DISCHARGE REED WITH A BACKING PLATE.

Fig. 3 shows a comparison of predicted and measured maximum strain along the centre line of a discharge reed. There was no backing plate in this case, but the reed tip was held on the seat (8). Using a finite element model of the reed with a coarse mesh (81 degrees of freedom for the whole reed) Fig. 4(a), and a point load at the centre of the port which varied according to the pressure-time history across the valve during the discharge phase of the compressor cycle, there was poor agreement between the predicted and measured maximum strain. With a finer mesh (114 degrees of freedom for half of the reed), Fig. 4(b), and distributed "consistent" loading over the port area the agreement improved. Forced vibration analyses always require much more computer resources than static problems (7). So in order to economise on computer resources the program was not run under dynamic operating conditions with the finer mesh and a distributed load. Instead, correction factors were developed by comparing the static displacement and stress patterns predicted by the coarse mesh and a point load with the static displacement and stress patterns predicted by the finer mesh and a distributed loading. These correction factors were then applied to the displacement and stress fields predicted by the coarse mesh under dynamic operating conditions. If adequate computer resources were available such a procedure would not be necessary.

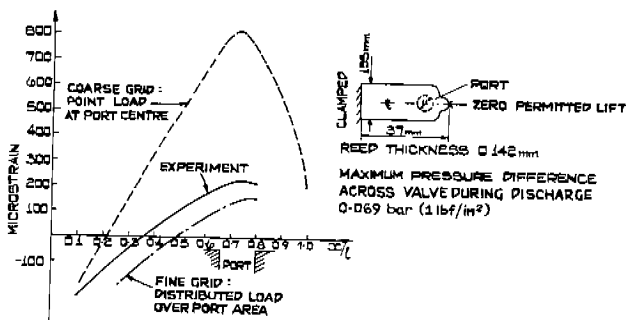


FIG. 3. DISTRIBUTION OF MAXIMUM STRAIN ALONG ϵ OF REED

It was concluded that a relatively fine grid and distributed loading over the valve port is necessary for reliable prediction of dynamic strain, particularly in the area of the port. However, a relatively coarse grid and a point load at the centre of the port can successfully predict dynamic displacements under operating conditions.

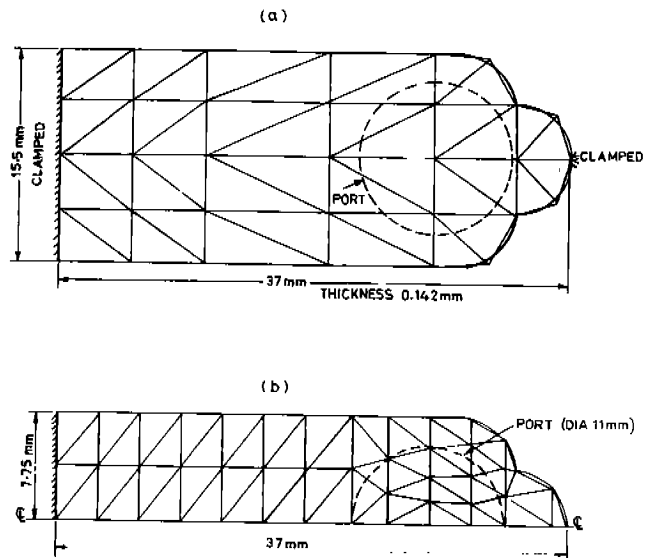


FIG. 4. FINITE ELEMENT MODEL OF CANTILEVER REED VALVE
(a) 85 DEGREES OF FREEDOM (WHOLE REED)
(b) 114 DEGREES OF FREEDOM (HALF REED)

The double differentiation of the predicted displacements to calculate strain is an inaccurate procedure. Errors can be very significant with complex geometries particularly when higher modes are excited. Under these conditions small discrepancies can involve large changes in curvature with resultant high stress levels.

CONCLUSIONS

Dynamic displacements of valve reeds and corresponding stresses under operating conditions are particularly important in relation to valve durability.

Computer programs, which applied the finite element method, predicted the dynamic displacement and stress patterns together with the natural frequencies and mode shapes of suction and discharge reeds. The non-linear boundary conditions at valve seat and stop (point or backing plate) were accounted for. There was acceptable agreement between predictions and measurements of displacements and strains under operating conditions. Provision can be made in the model to account for imperfect clamping at the reed root (7).

A relatively fine grid, distributed loading over the port area and a sufficiently small time increment in the integration scheme employed are necessary to predict dynamic stresses in reeds under operating conditions particularly when the reed strikes the valve seat or stop.

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